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OPTIMIZATION OF CONDENSING TEMPERATURE FOR NUCLEAR TURBOELECTRIC SPACE POWER PLANT

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ABSTRACT

In order to scope out operating parameters for a nuclear electric power plant, a relatively simple scheme has been derived for selecting the ratio of radiator condensing temperature to turbine inlet temperature. The optimum condensing temperature satisfies a set of approximate relations which minimize overall powerplant specific weight for any given electrical power output with respect to the fraction of plant weight which is radiator. As with all such analytical attempts, the validity of the results are contingent on the accuracy of the simplifying assumptions necessary to mathematically formulate the problem. These assumptions are presented and discussed in this Report.

The analysis indicates that over the range of anticipated radiator weight fractions and deviations from Carnot cycle efficiency, a choice of condensing to saturated turbine inlet temperature ratio of 0.60 to 0.70 is warranted. The analysis does not consider any weight penalties associated with higher vapor specific volumes or condensate pumping problems at lower condensing temperatures.

I. ASSUMPTIONS

The system analyzed is a simple two loop turbogenerator plant as shown in Fig. 1.

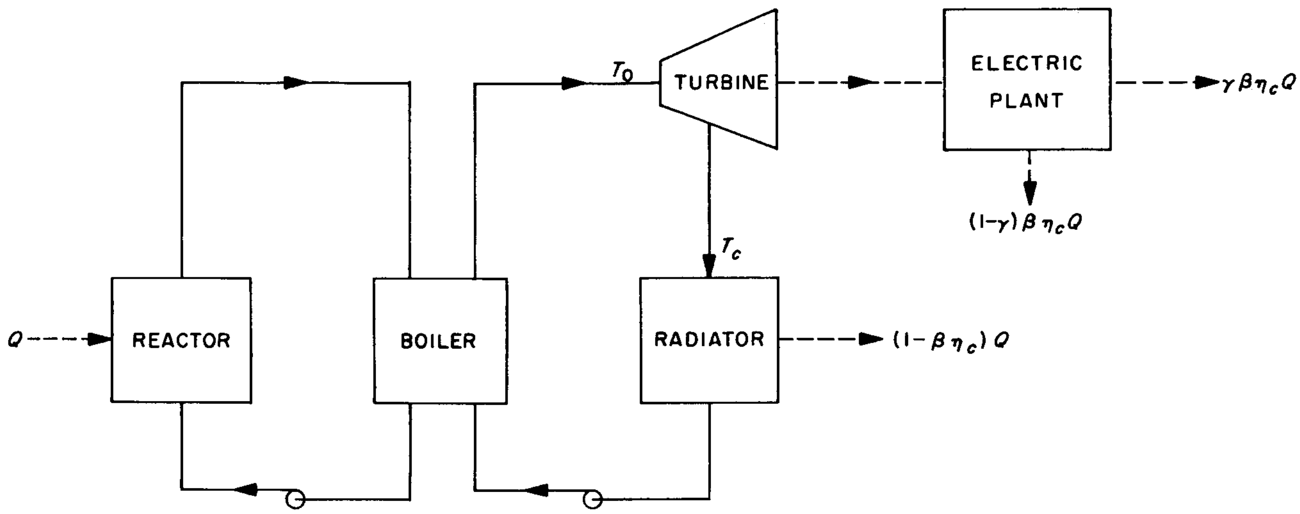

Fig. 1. Plant schematic

Assumption A. Power plant weight, up to and including the turbine, is directly proportional to heat produced and independent of condensing temperature.

For a given set of engineering design requirements, e.g., core power density and heat flux, boiler heat flux, coolant core temperature rise, etc., this assumption is reasonable for the reactor core and boiler, and a fair approximation for primary and secondary system plumbing. Items which are independent of power such as control drive mechanisms, control system, etc., may be estimated and subtracted from the total power plant weight which is heat output dependent.

Turbine weight is dependent on heat output, although probably not in a linear fashion, and is also dependent on condensing temperature to the extent that exhaust specific volume will influence final stage design. Since the turbine represents a relatively small portion of the plant weight, lack of inclusion of this temperature dependence is probably not of too great a significance.

Perhaps the greatest error which can result from this assumption is shield weight variation with heat output. If reactor weight increases proportional to heat output, shield weight in a given design will probably vary more slowly due to the exponential nature of the attenuation process controlling the thickness while geometric considerations control the diameter.

Assumption B. Generator, power conditioning equipment, ion engine, and auxiliary radiator weights are directly proportional to electrical power.

Since individual component efficiencies in this group of equipment are invariant for this optimization study, they may all be expressed as directly proportional to ion beam power. The auxiliary radiator is assumed to radiate at some fixed temperature, invariant in this optimization.

Assumption C. Radiator weight is directly proportional to heat rejected and inversely proportional to the fourth power of the rejection temperature.

This assumption is open to question from three major areas:

1. Emissivity of metals in the higher temperature ranges are generally changing with temperature, and the actual weight-temperature dependence will be different than quadric.
2. If a radiator is designed to meet a certain *whole radiator* meteorite penetration probability in a given time, then absolute size will influence meteorite protection per unit area, and the implicit assumption of fixed weight per unit area at a given condensing temperature is not strictly valid. On the other hand, if penetration probability per unit time *per unit area* is a criterion via the introduction of redundant design and a means for detection and isolation of a leaking tube or section, the assumption of fixed weight per unit area is approached.
3. Radiator weight per unit area will generally increase with decreasing condensing temperature due to increasing tube size for the higher specific volume vapor in addition to the quadric temperature dependence. The optimum distribution between tube and fin weights is also temperature dependent.

Depending on specific radiator design, meteorite penetration requirements, and operating parameters, the weight per unit area – condensing temperature variation can probably range from T^{-3} to T^{-6}. T^{-4} is arbitrarily chosen for this analysis. If the overall weight-temperature relation can be expressed as T_c^{-m} where m is reasonably constant over the range of interest, the analysis can readily be modified to take this into account.

Assumption D. Gross turbine efficiency may be expressed as some constant times Carnot efficiency.

This assumption is a fairly reasonable approximation. The constant represents the turbine deviation from Carnot efficiency brought about by non-isentropic expansion, Rankine to Carnot cycle differences, and superheat, if superheated vapor is utilized. It should be noted that Carnot cycle inlet temperature should be taken as the saturation temperature corresponding to the turbine inlet pressure. Slight improvement in efficiency due to use of superheated vapor is taken into account in the constant for best accuracy of this representation. The "constant" actually shows some variation with exhaust temperature primarily due to the effect of moisture content on the deviation of the expansion from isentropic behavior. This effect is sometimes estimated as 1% additional efficiency loss per 1% additional moisture. (The validity of even this estimate is held questionable.) In any case, stage moisture separation would tend to improve the accuracy of this representation.

Assumption E. Beam power may be expressed as some constant times gross electric output.

At a given ion motor specific impulse this is felt to be a reasonable assumption. While feedback electrical loads for pumps, etc., may show some very minor variations with plant efficiency due to changing heat output for a given electric output, this constitutes so small a variation as to be generally negligible.

II. ANALYSIS

A block diagram is shown in Fig. 2, separating the plant into the three basic weight controlled elements of heat plant, radiator, and electric plant.

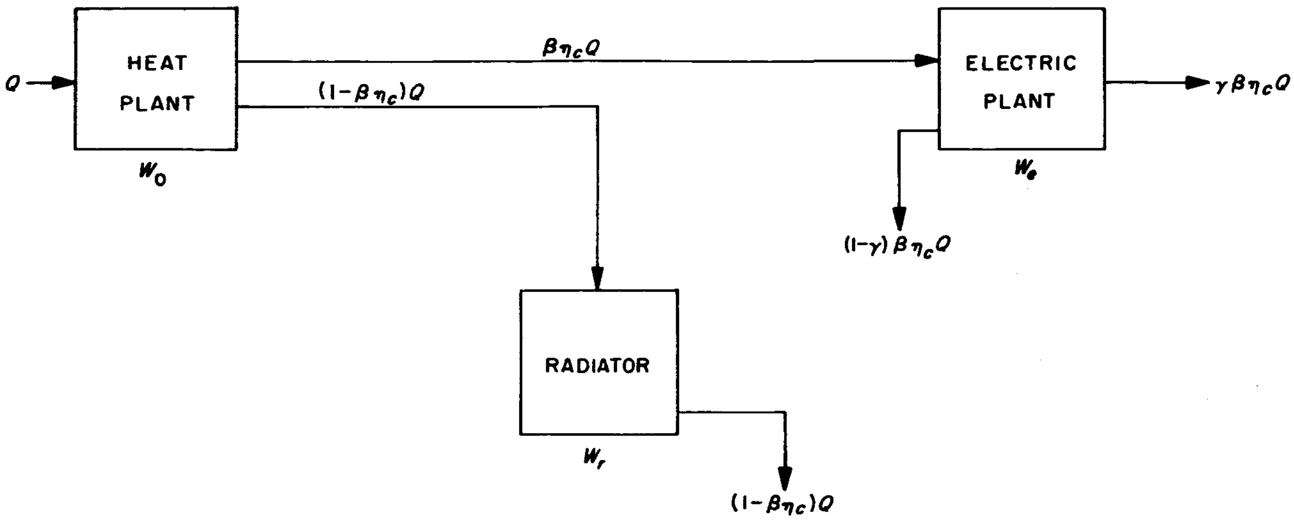

Fig. 2. Weight elements

By assumption IA:

$$W_0 = K_0 Q \tag{1}$$

By assumption IB:

$$W_e = K_e P \tag{2}$$

By assumption IC:

$$W_r = K_r \left(\frac{T_c^0}{T_c}\right)^4 (1 - \beta\,\eta_c)\,Q \tag{3}$$

where T_c^0 is some convenient reference temperature utilized for the calculation of K_r. For a given radiator mechanical design $K_r T_c^{04}$ would be fixed and independent of actual condensing temperature T_c. The only incentive for expressing K_r in this fashion is to maintain consistent units for all the K's.

By assumption ID:

$$\eta_t = \beta\eta_c = \beta \frac{T_0 - T_c}{T_0} \text{ (gross turbine efficiency)} \tag{4}$$

By assumption IE:

$$P = \gamma\eta_t Q = \gamma\beta \frac{T_0 - T_c}{T_0} Q \text{ (beam power)} \tag{5}$$

Total plant weight, combining Eq. (1), (2), and (3) and substituting for Q from Eq. (5):

$$W_T = \frac{K_0 P}{\gamma\beta} \frac{T_0}{T_0 - T_c} + K_e P + \frac{K_r P}{\gamma\beta} \left(\frac{T_c^0}{T_c}\right)^4 \left(1 - \beta \frac{T_0}{T_0 - T_c}\right) \frac{T_0}{T_0 - T_c}$$

or

$$\frac{W_T}{P} = \frac{K_0}{\gamma\beta} \frac{T_0}{T_0 - T_c} + K_e + \frac{K_r}{\gamma\beta} \left(\frac{T_c^0}{T_c}\right)^4 \left(1 - \frac{\beta T_0}{T_0 - T_c}\right) \frac{T_0}{T_0 - T_c} \tag{6}$$

Note that W_T/P represents the overall power plant specific weight related to beam electric power. Differentiation of Eq. (6) with respect to T_c and equating to zero will minimize this specific weight. It should be further noted that the minimum specific weight occurs at the same T_c independent of the electric power basis used in the definition of specific weight. Equation (6) may be divided by γ thereby expressing specific weight in terms of input power to the electric plant without changing the optimum T_c. Differentiating, equating to zero, and collecting terms yields:

$$\frac{K_0}{K_r} \left(\frac{T_c}{T_c^0}\right)^4 = 4 \frac{T_0}{T_c} (1 - \beta) - 4\beta \frac{T_c}{T_0} + 8\beta - 5 \tag{7}$$

Note that the electric plant weight parameter K_e and the electric plant efficiency γ have dropped out of the optimization. Letting $X = T_c/T_0$ condensing to inlet saturated temperature ratio:

$$\frac{K_0}{K_r}\left(\frac{T_0}{T_c^0}\right)^4 X^4 = \frac{4(1-\beta)}{X} - 4\beta X + 8\beta - 5 \tag{8}$$

The optimum temperature ratio X is seen to be a function of the turbine performance parameter β, turbine saturated inlet temperature T_0, and the weight parameters for the heat plant K_0, and radiator K_r calculated at some arbitrary reference temperature T_c^0 (noting that $K_r T_c^{0^4}$ should be a constant for a given radiator mechanical design).

In an attempt to present the results graphically in perhaps a more readily apparent physical significance, two additional steps have been taken. First, recall that the reference condensing temperature T_c^0 for calculation of K_r is perfectly arbitrary. In plotting Fig. 3, T_c^0 is *chosen* as 3/4 T_0, thereby substituting a number for the temperature ratio T_0/T_c^0 in Eq. (8). Use of the graph therefore necessitates the calculation of a reference radiator weight at a condensing temperature of 3/4 T_0 for purposes of determining the *optimum* condensing temperature. (The choise of 3/4 T_0 is made on the basis that minimum *radiator* weight occurs at this condensing temperature, assuming constant emissivity, area weight proportionality, and Carnot cycle efficiency ($\beta = 1$) and is many times the initial selection of condensing temperature for scoping out radiator weights.) In this fashion, a trial-and-error approach may be completely avoided. The additional step is to plot X versus $K_r/K_r + K_0$ rather than the ratio K_0/K_r. $K_r/K_r + K_0$ may be defined as the fraction of powerplant weight (excluding the entire electric plant and auxiliary radiator, i.e., everything "downstream" of the turbine), which is reference radiator *where the reference radiator weight is calculated at a condensing temperature three-fourths of the saturated inlet turbine temperature.* [It is the hope of the writer that this in fact does lend more physical significance to the reader's interpretation of the graphical results than merely plotting X vs $K_0/K_r\ (T_0/T_c^0)^4$]. The results are presented for three values of β which should cover in most cases the expected range of variation of this turbine performance parameter.

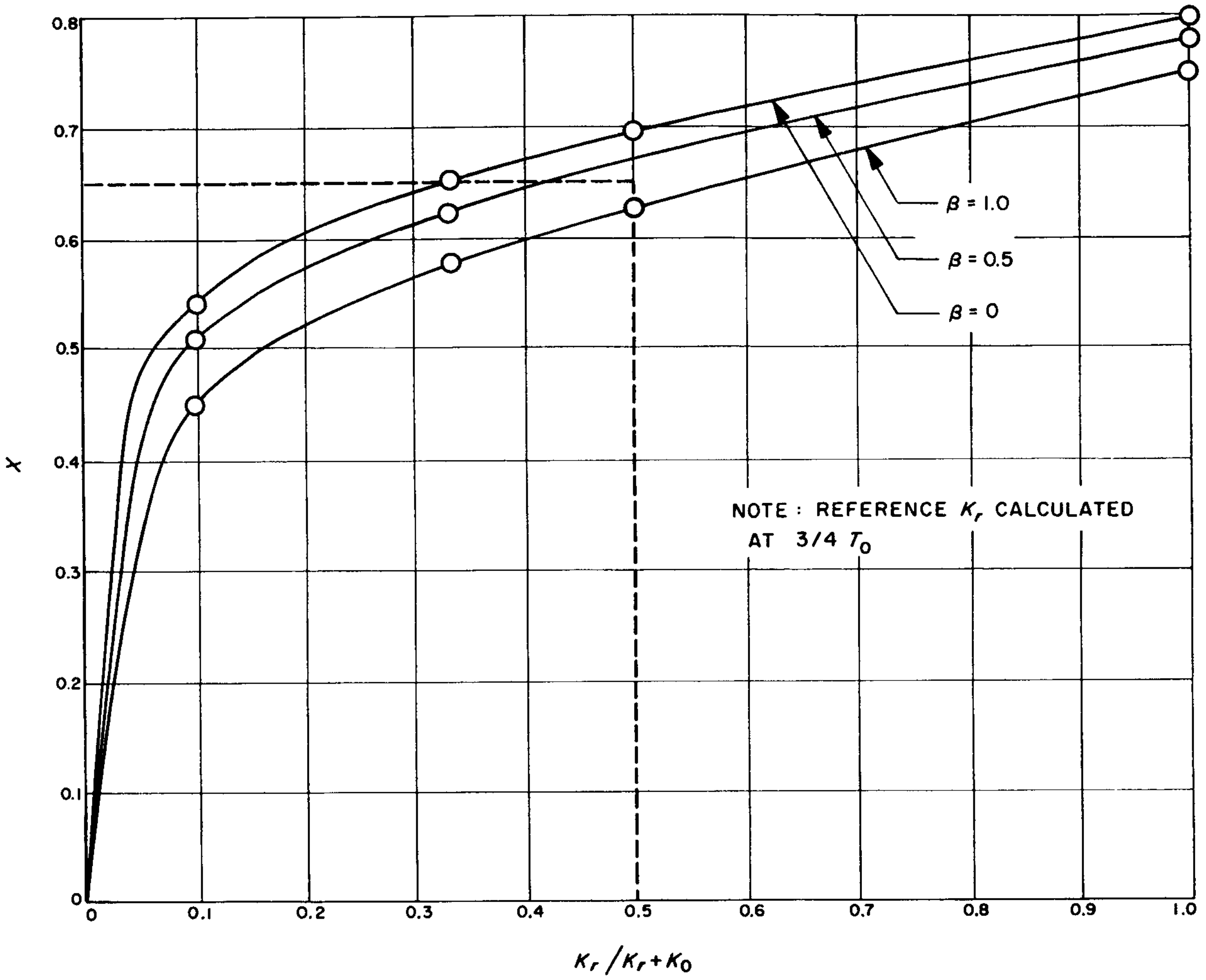

Fig. 3. Optimum condensing to saturated turbine inlet temperature vs reference radiation weight fraction

III. DISCUSSION OF RESULTS

It is seen that the optimum temperature ratio is relatively insensitive to β and a value for this parameter from 0.6 to 0.9 should cover most practical turbine designs with superheat values from zero to several hundred degrees and exit moisture contents from 5% to 15%. The expected range of variation of $K_r/K_0 + K_r$ depending on specific radiator design and meteorite penetration criteria is perhaps from 1/3 to 2/3; 1/3 being an unusually light radiator and 2/3 being a fairly heavy radiator. Over this range of $K_r/K_0 + K_r$ and β variation the optimum temperature ratio ranges from about 0.60 to 0.70. A good mean value, for $K_r/K_0 + K_r = 0.5$ and $\beta = 0.75$, is $X = 0.65$.

The difference between $X = 0.65$ based on the above system and $X = 0.75$ based strictly on radiator optimization (for $\beta = 1$) is quite a significant difference in condensing temperature. For example, a saturated turbine inlet temperature of 1940°F (2400°R) would result in an optimum condensing temperature of 1100°F (1560°R) for the whole plant optimization and 1340°F (1800°R) based on the simple 3/4 rule. With potassium as a working fluid 1100°F results in a vapor pressure of only 2 psia and requires further study of the weight penalties which might be incurred due to the high vapor specific volumes and pumping requirements of the liquid condensate.

The results of this analysis may serve as a starting point to grossly scope out the range of interest for optimum condensing temperature at the initial stages of cycle analysis or plant design investigations for nuclear turboelectric spacecraft.

NOMENCLATURE

K_e electric plant weight per unit beam power

K_0 heat plant weight per unit heat output

K_r radiator weight per unit heat rejected at temperature T_c°

P beam power

Q heat output

T_c radiator condensing temperature

T_0 turbine saturated inlet temperature corresponding to inlet pressure

T_c° reference radiator condensing temperature

W_e electric plant weight

W_0 heat plant weight

W_r radiator weight

W_T sum of electric plant, heat plant, and radiator weight

X optimum ratio of condensing temperature to turbine saturated inlet temperature

β ratio of gross turbine efficiency to Carnot efficiency

γ ratio of beam power to gross turbine output power

η_c Carnot efficiency

η_t gross turbine efficiency